

# HYBRID SEVEN-BAR PRESS MECHANISM: LINK OPTIMIZATION AND KINETOSTATIC ANALYSIS

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**Abstract:** An optimization study with kinetostatic analysis is performed on hybrid seven-bar press mechanism. This study is based on previous studies performed on planar hybrid seven-bar linkage. Dimensional synthesis is performed, and optimum link lengths for the mechanism are found. Optimization study is performed by using genetic algorithm (GA). Genetic Algorithm Toolbox is used with Optimization Toolbox in MATLAB®. The design variables and the constraints are used during design optimization. The objective function is determined and eight precision points are used. A seven-bar linkage system with two degrees of freedom is chosen as an example. Metal stamping operation with a dwell is taken as the case study. Having completed optimization, the kinetostatic analysis is performed. All forces on the links and the crank torques are calculated on the hybrid system with the optimized link lengths.

**Keywords:** hybrid mechanism; genetic algorithm; kinetostatic analysis; linkage optimization; mechanism synthesis; seven link press mechanism

## 1 INTRODUCTION

Many industrial applications are potentially used as hybrid mechanisms. Some of them can be used for cutting, printing, and stamping. Hybrid mechanism configurations are seen in literature: five-bar slider crank, and seven-bar, also nine-bar. Two different actuators are combined with a mechanism to give it a programmable motion. A constant velocity (CV) motor and servo motor (SM) are the actuators. However, while introducing different configurations, there is a need for synthesis to get the links lengths as its optimum [1].

Some studies have given kinematics and dynamics of hybrid configurations. Here some of them are taken as the optimization methods for synthesis of a seven-bar press mechanism. Simulated annealing (SA), genetic algorithms (GA), neural networks (NN), particle swarm optimization (PSO), and the others are applied in studies [2]. M. A. Laribi et al. [3] have used a hybrid method GA-FL for solving the problem of path generation in synthesis of a four-bar mechanism. Blackett [4] has performed synthesis of a five-bar mechanism. The Hooke and Rees pattern search technique has been applied. R. McDougall [5] has used Optimization based mechanism synthesis (OBMS) using four-bar and five-bar mechanisms using PSO algorithm. C. Meng et al. [6] have presented a research on a press introducing hybrid input. Inverse kinematics is studied for hybrid driven mechanism. Z. Yuan et al. [7] have studied two different hybrid mechanisms to get an output motion by minimizing torque and power from the servo motor. Their arrangement is called the slider hybrid mechanism. K. Zang [8] has studied hybrid mechanism with a five-bar. A hybrid five-bar configuration is presented with related kinematics and its mathematical model. Li and Zhang [9] have used a seven-bar mechanism press with hybrid driven case for deep drawing. M. E. Kütük [10] has performed a study on inverse kinematics issues on hybrid driven seven-bar mechanism with motion design for press application. The dimensions of

a seven-bar hybrid press mechanism are found by using experimental optimization. S. Ebrahimi and P. Payvandy [11] have presented a study on the dimensional synthesis using different heuristic optimization techniques: GA, PSO and DE.

This study is based on an industrial project for a press mechanism designed with a seven-bar mechanism. Motion with a dwell segment is given for a deep drawing study. MATLAB® GA solver is used in Global Optimization Toolbox. Synthesis of a seven-bar mechanism is presented by using eight precision points. Having performed optimization, the mechanism is drawn. Then kinetostatic analysis is achieved by applying a representative force on the ram. The results are presented and discussed.

## 2 SEVEN-BAR PRESS MECHANISM

A seven-bar mechanism is used for producing different ram motion scenarios. A hybrid press mechanism is given a detailed motion design procedure in Kütük's study [1]. Forward and inverse kinematics issues are explicitly given. One axis is driven by a CV motor and the other axis is driven by an SM. The seven-bar press mechanism referred is shown below in Fig. 1.

Loops 1 and 2:

$$\vec{r}_5 + \vec{r}_6 = \vec{FG} + \vec{GC} + \vec{r}_4 \quad (1)$$

$$\vec{r}_5 + \vec{r}_6 = \vec{r}_1 + \vec{r}_2 + \vec{r}_3 \quad (2)$$

Here  $r_1$ ,  $r_2$ ,  $r_3$ ,  $r_4$ ,  $r_5$ ,  $r_6$ ,  $\theta$  and  $e$  are described as the dimensional parameters of the seven-bar press mechanism. The slider motion ( $s$ ) is expressed as  $s = f(\theta_2, \theta_5)$ . The angular displacement, angular velocity and angular acceleration of link  $r_5$  are driven by a CV motor. The angular displacement, angular velocity and angular acceleration of link  $r_2$  are driven by an SM.

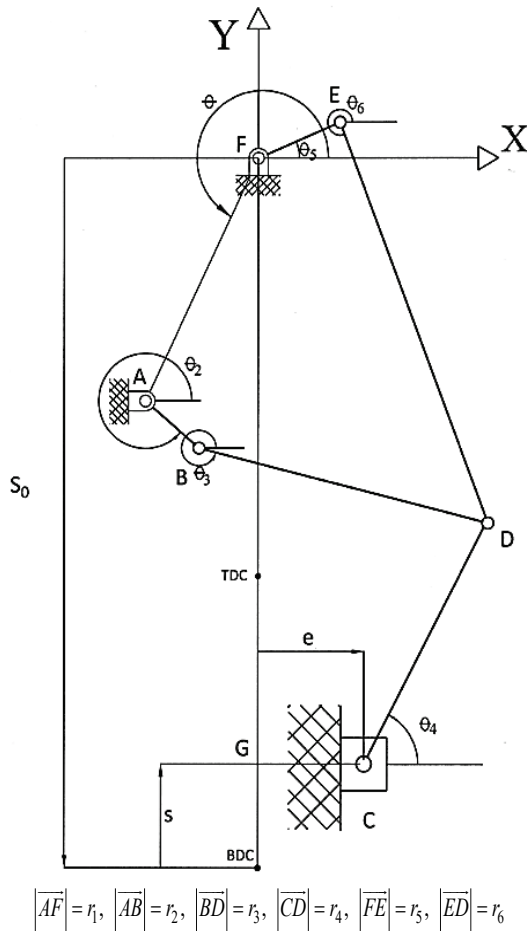


Figure 1 Seven-bar mechanism [1]

## 2.1 Synthesis of a Seven-Bar Mechanism

Li et al. [12] have given the classifications of hybrid five-bar linkages by using the unrestraint double-crank type. Kütük et al. have then performed an experimental optimization [10, 13]. A traditional trial and error approach is used to design a seven-bar press mechanism. The input link  $r_2$  and  $r_5$  are given as unrestraint cranks.

$$r_1 + r_2 + r_5 < r_3 + r_6 \quad (3)$$

$$r_2 + r_5 + r_6 < r_1 + r_3 \quad (4)$$

$$r_2 + r_3 + r_5 < r_1 + r_6 \quad (5)$$

The inequality constraints should be satisfied to provide slider mobility condition.  $X_D$  gives the coordinate of point D in  $x$  direction in Eq. (6). The link lengths inequality constraints are given in Eq. (7).  $S_0$  is the point where  $r_5$ ,  $r_6$  and  $r_4$  are lined up in a straight line. By using Eq. (8);

$$r_4 > X_D - e \quad (6)$$

$$r_i \min \leq r_i \leq r_i \max \quad (7)$$

$$r_i \min \leq r_i \leq r_i \max \quad (8)$$

This study was started by using the values found previously when deciding intervals during optimization. M.E. Kütük found them while performing conceptual design during his M.Sc. thesis [1] as:  $r_1 = 530$  mm,  $r_2 = 200$  mm,  $r_3 = 650$  mm,  $r_4 = 900$  mm,  $r_5 = 170$  mm,  $r_6 = 800$  mm,  $e = 6.73$  mm.

## 3 MOTION EXAMPLE

The displacement of the slider link is given in Fig. 2(a). Stroke of the slider is 687 mm with 10 stroke/minute and the motion has got 3 segments. The motion kinematics is illustrated in Tab. 1. The segment number, time interval and initial and final values of position, velocity and acceleration of each segment are given.

Table 1 Kinematics of the dwell type motion

Segment number	Motion time (s)	Position (mm)	Velocity (mm/s)	Acceleration (mm/s <sup>2</sup> )
1	$0 \leq t \leq 3$	687.0	0.0	0.0
2	$3 \leq t \leq 3.5$	0.0	0.0	0.0
3	$3.5 \leq t \leq 6$	0.687	0.0	0.0

The 5<sup>th</sup> link is driven by CV motor. It rotates  $2\pi$  radians in counterclockwise (CCW) direction. Its angular velocity is a constant value, 1.0472 rad/s in Fig. 2(b). The red circles in Fig. 2(a) and (b) show eight precision points taken during dimensional synthesis.

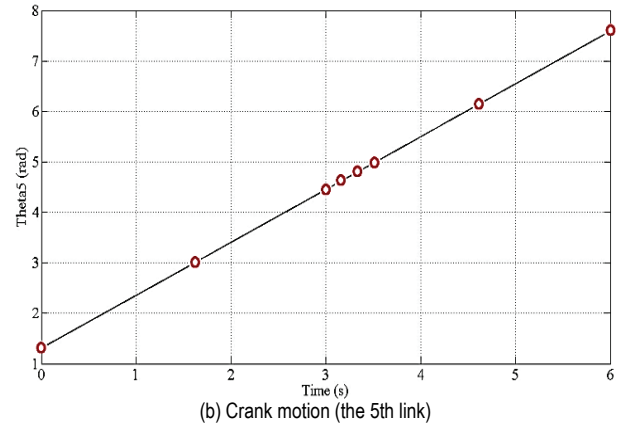
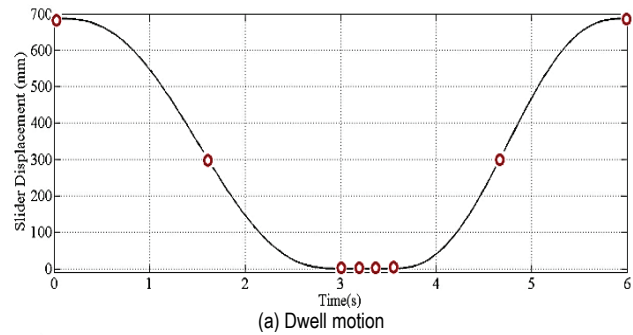


Figure 2 Slider and crank motion

## 4 OPTIMIZATION PROBLEM WITH GENETIC ALGORITHM

The mechanism optimization problem is separated into four parts: the conceptual design; the model development with

design variables; building of an objective function; and finding the best values satisfying the design constraints for the mechanism [14]. Optimization is very important in engineering design and can solve many synthesis problems. The link lengths and initial angles describe the design variables of the mechanism. The constrained nonlinear optimization is performed for a seven-bar hybrid mechanism. The optimization objective function and the constraints are determined by considering the press motion characteristics. The following optimization problem represents the constrained nonlinear optimization. Objective function is described first. Forward kinematics equations are used to get the objective function. The objective function is given in Eq. (9) by minimizing  $f_{obj}$  as:

$$f_{obj} = \sum_{i=1}^k (s_d^i - s_g^i)^2 \quad (9)$$

where  $s_d^i$  represents the design precision points required, and  $s_g^i$  is the given point for the ram. Here eight precision points are taken. The results are obtained having run the optimization program many times. The design variables are:

$$X = [r_1, r_2, r_3, r_4, r_5, r_6, e, \theta, \theta_{2i}] \quad (10)$$

where  $\theta_{2i}$  is given as eight points from  $i = 1, \dots, 8$ . Lower and upper limits of a seven-bar mechanism are defined as: [500, 150, 600, 850, 150, 750, 0,  $\pi$ , 0, 0, 0, 0, 0, 0, 0] and [600, 250, 700, 950, 250, 850, 10,  $3\pi/2$ ,  $2\pi$ ,  $2\pi$ ,  $2\pi$ ,  $2\pi$ ,  $2\pi$ ,  $2\pi$ ,  $2\pi$ ].

They are set while performing the optimization algorithm in Matlab [15]. GA Toolbox is applied for getting optimum values for the links. GA is set as follows: number of generation  $NP = 100$ , Crossover probability  $CP = 0.6$ , Adaptive feasible, Mutation probability  $MP = 0.1$ , roulette wheel is chosen referring to previous studies performed. The synthesis study is being started by using Return-TDC as the first precision point, Return-intermediate as the second precision point, four points during dwell at BDC, one point for Rise-intermediate, and one point for Rise-TDC. Optimum solution performing eight precision points are determined. The objective function values are found between  $1.23 \times 10^{-5}$  and  $-6.99 \times 10^{-8}$ . The seven link lengths and the offset are taken as:  $r_1 = 571.77$  mm,  $r_2 = 177.99$  mm,  $r_3 = 656.53$  mm,  $r_4 = 853.96$  mm,  $r_5 = 198.94$  mm,  $r_6 = 756.89$  mm,  $e = 7.38$  mm.

## 5 HYBRID SEVEN-BAR MECHANISM IN MATLAB®

A program based on forward kinematics equations is written in Matlab® in order to see the simultaneous positions of the hybrid seven-bar mechanism. The angular positions of the second and the fifth links must be given as the input values. The hybrid system with eight different values of  $\theta_2$  and  $\theta_5$  are illustrated in Figs. 3-5. The hybrid mechanism tracing precision points are given in Fig. 2(a). Here  $s = 0$  and  $s = 687$  mm are taken by referring to study [1]. Industrial applications will refer

to dwell period for the second segment of the motion, which is very important for metal forming operations.

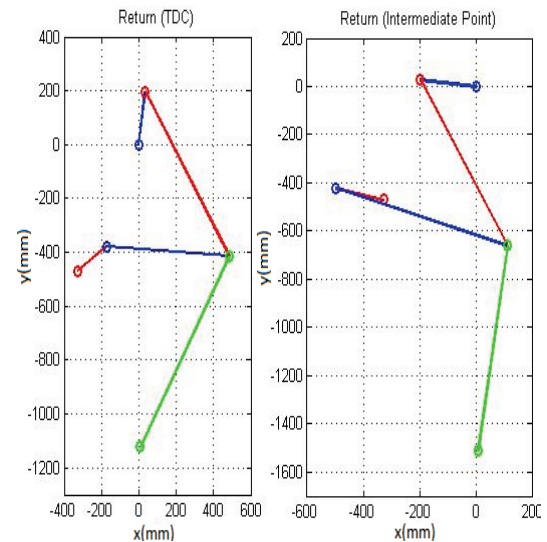


Figure 3 TDC and intermediate point in return period

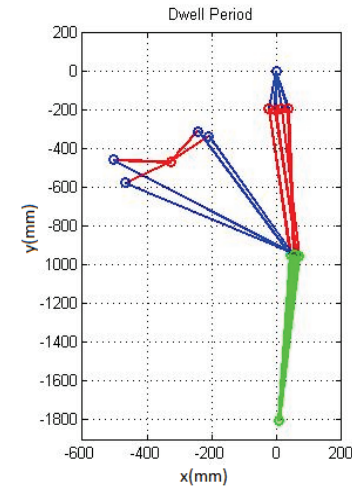


Figure 4 Dwell period (BDC-4 precision points)

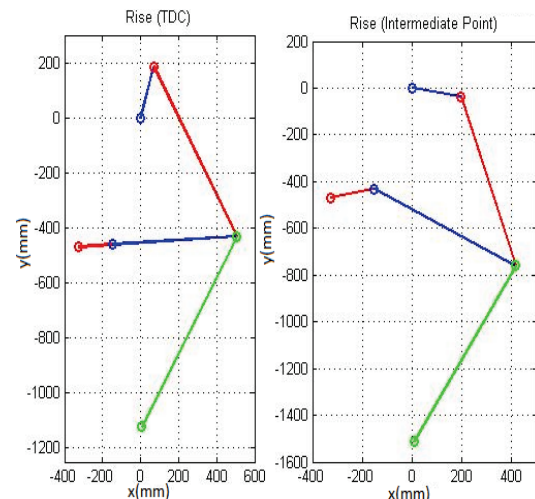


Figure 5 TDC and intermediate point in rise period

## 6 THE DYNAMIC MODEL FROM KINETO-STATIC METHOD

The kinetostatic method is also known as D'Alembert technique. This method rearranges the Newton's second law. A dynamic problem is transformed into a static one by adding fictitious inertial forces and inertial torques onto the system. One fictitious force on each moving body which is equal to the mass of that body times the acceleration of its mass center. The direction is opposite to its acceleration. It is directly applied onto the center of gravity, apart from the already existing real forces. One fictitious torque on each moving body which is equal to the centroidal inertia of that body times its angular acceleration. The direction is opposite to that of acceleration. The already existing torque is also applied. The model is then analyzed with the rules of statics [16]. The free body diagrams of the links are given in Fig. 6.

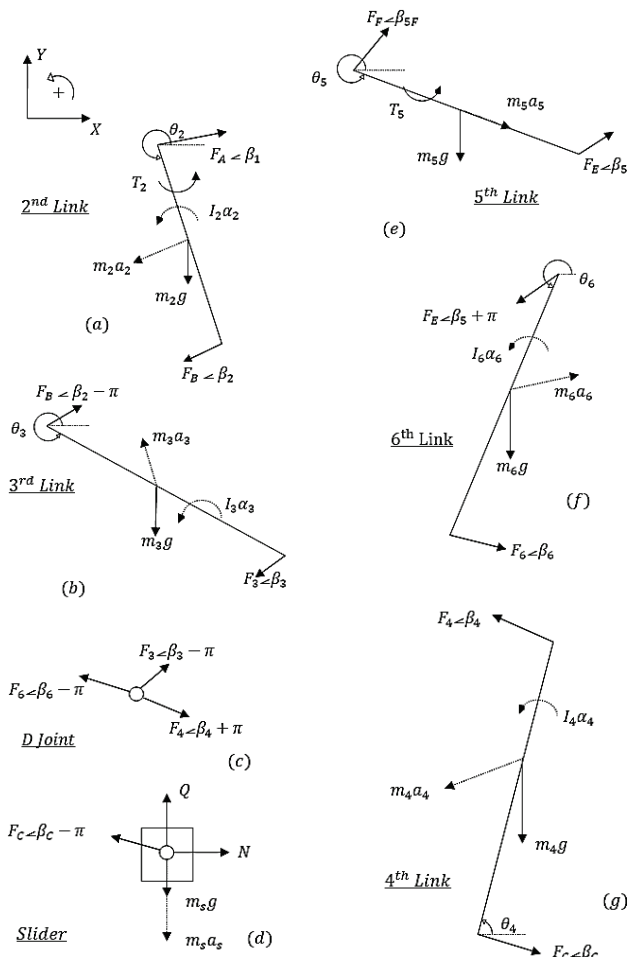


Figure 6 Free body diagram of the links

In the dynamic model, the velocity fluctuation of the constant velocity motor, the elastic deformation and friction force of the links are assumed as zero to make the system simple. Kinetostatic diagram of each link in hybrid mechanism is shown in Fig. 6 (a, b, c, d, e, f, g). Inertia forces and inertia torques are acted on every link of the hybrid driven mechanism. They are shown by dashed vectors. Thus, the static equilibrium equations of each link can be constructed [17]. The links used

in the mechanism are taken as solid cylinders. The parameters of the links are given in Tab. 2.

Table 2 The parameters of the links

Parameters	Link 2	Link 3	Link 4	Link 5	Link 6	Slider
Radius (m)	0.025	0.025	0.025	0.025	0.025	-
Mass (kg)	3	9.95	13.78	2.6	12.25	20
Moment of inertia (kg·m <sup>2</sup> )	0.0105	0.3519	0.9323	0.0067	0.6552	-

### Link AB

$$F_{Ax} - F_{Bx} = m_2 \ddot{X}_{cg2} \quad (11)$$

$$F_{Ay} - F_{By} - m_2 g = m_2 \ddot{Y}_{cg2} \quad (12)$$

$$r_2' = [r_{Acg2} \cos(\theta_2 - \pi), r_{Acg2} \sin(\theta_2 - \pi), 0] \quad (13)$$

$$F_A = [F_{Ax}, F_{Ay}, 0] \quad (14)$$

where  $r_{Acg2}$  is the distance between the mass center of the second link and A point.

$$r_2'' = [r_{Bcg2} \cos \theta_2, r_{Bcg2} \sin \theta_2, 0] \quad (15)$$

$$F_B = [-F_{Bx}, -F_{By}, 0] \quad (16)$$

where  $r_{Bcg2}$  is the distance between the mass center of the second link and B point.

$$\sum M_{cg2} = (r_2' \times F_A) + (r_2'' \times F_B) + T_2 = I_2 \ddot{\theta}_{cg2} \quad (17)$$

### Link BD

$$F_{Bx} - F_{3x} = m_3 \ddot{X}_{cg3} \quad (18)$$

$$F_{By} - F_{3y} - m_3 g = m_3 \ddot{Y}_{cg3} \quad (19)$$

$$r_3' = [r_{Bcg3} \cos(\theta_3 - \pi), r_{Bcg3} \sin(\theta_3 - \pi), 0] \quad (20)$$

$$F_B = [F_{Bx}, F_{By}, 0] \quad (21)$$

where  $r_{Bcg3}$  is the distance between the mass center of the third link and B point.

$$r_3'' = [r_{Dcg3} \cos \theta_3, r_{Dcg3} \sin \theta_3, 0] \quad (22)$$

$$F_D = [-F_{3x}, -F_{3y}, 0] \quad (23)$$

where  $r_{Dcg3}$  is the distance between the mass center of the third link and D point.

$$\sum M_{cg_3} = (r_3' \times F_B) + (r_3'' \times FD) = I_3 \ddot{\theta}_{cg_3} \quad (24)$$

Link DC

$$-F_{4x} + F_{C_x} = m_4 \ddot{X}_{cg4} \quad (25)$$

$$F_{4_y} - F_{C_y} - m_4 g = m_4 \ddot{Y}_{cg4} \quad (26)$$

$$r_4' = [r_{Dcg4} \cos \theta_4, r_{Dcg4} \sin \theta_4, 0] \quad (27)$$

$$F_D = [-F_{4_x}, F_{4_y}, 0] \quad (28)$$

where  $r_{Dcg_4}$  is the distance between the mass center of the fourth link and D point.

$$r_4'' = [r_{Ccg_4} \cos(\theta_4 + \pi), r_{Ccg_4} \sin(\theta_4 + \pi), 0] \quad (29)$$

$$F_C = [F_{C_x}, -F_{C_y}, 0] \quad (30)$$

where  $r_{Ccg4}$  is the distance between the mass center of the fourth link and C point.

$$\sum M_{cg4} = (r_4' \times F_D) + (r_4'' \times F_C) = I_4 \ddot{\theta}_{cg4} \quad (31)$$

EF Link

$$F_{F_x} + F_{E_x} = m_5 \ddot{X}_{cg5} \quad (32)$$

$$F_{F_v} - F_{E_v} - m_5 g = m_5 \ddot{Y}_{cg5} \quad (33)$$

$$r_5' = [r_{Fcgs} \cos(\theta_5 - \pi), r_{Fcgs} \sin(\theta_5 - \pi), 0] \quad (34)$$

$$F_F = [F_{F_x}, F_{F_y}, 0] \quad (35)$$

where  $r_{Fcg5}$  is the distance between the mass center of the fifth link and F point.

$$r_5'' = [r_{Ecgs} \cos \theta_5, r_{Ecgs} \sin \theta_5, 0] \quad (36)$$

$$F_E = [F_{E_x}, F_{E_y}, 0] \quad (37)$$

where  $r_{Ecg5}$  is the distance between the mass center of the fifth link and E point.

$$\sum M_{cg5} = (r_5' \times F_F) + (r_5'' \times F_E) + T_5 = I_5 \ddot{\theta}_{cg5} \quad (38)$$

ED Link

$$-F_{E_r} + F_{6_r} = m_6 \ddot{X}_{cg6} \quad (39)$$

$$-F_{E_v} - F_{6_v} - m_6 g = m_6 \ddot{Y}_{cg6} \quad (40)$$

$$r_6' = [r_{Ec g_6} \cos(\theta_6 - \pi), r_{Ec g_6} \sin(\theta_6 - \pi), 0] \quad (41)$$

$$F_E = [-F_{E_x}, -F_{E_y}, 0] \quad (42)$$

where  $r_{Ecg6}$  is the distance between the mass center of the sixth link and E point.

$$r_6'' = [r_{Dcg_6} \cos \theta_6, r_{Dcg_6} \sin \theta_6, 0] \quad (43)$$

$$F_D = [F_{6x}, -F_{6y}, 0] \quad (44)$$

where  $r_{Dcg_6}$  is the distance between the mass center of the sixth link and D point.

$$\sum M_{cg6} = (r_6' \times F_E) + (r_6'' \times F_D) = I_6 \ddot{\theta}_{cg6} \quad (45)$$

### D Joint

$$F_{3_r} + F_{4_r} - F_{6_r} = 0 \quad (46)$$

$$F_{3_v} - F_{4_v} + F_{6_v} = 0 \quad (47)$$

### Slider

$$-F_{C_r} + N = 0 \quad (48)$$

$$Q + F_{C_y} - m_{slider}g = m_{slider}a_{slider} \quad (49)$$

The static equilibrium equations are given above. Nineteen equations are obtained with nineteen unknowns. These equations are linear sets and they can be expressed in matrix form as

$$DE=F \tag{50}$$

where

[illegible]

$$E = \begin{bmatrix} F_{A_x}, F_{A_y}, F_{B_x}, F_{B_y}, F_{3_x}, F_{3_y}, F_{4_x}, F_{4_y}, F_{C_x}, \\ F_{C_y}, F_{F_x}, F_{F_y}, F_{E_x}, F_{E_y}, F_{6_x}, F_{6_y}, T_2, T_5, N \end{bmatrix}^T$$

$$F = \begin{bmatrix} m_2 \ddot{X}_{cg2}, m_2 \ddot{Y}_{cg2} + m_2 g, I_2 \ddot{\theta}_{cg2}, m_3 \ddot{X}_{cg3}, m_3 \ddot{Y}_{cg3} + m_3 g, I_3 \ddot{\theta}_{cg3}, m_4 \ddot{X}_{cg4}, \\ m_4 \ddot{Y}_{cg4} + m_4 g, I_4 \ddot{\theta}_{cg4}, m_5 \ddot{X}_{cg5}, m_5 \ddot{Y}_{cg5} + m_5 g, I_5 \ddot{\theta}_{cg5}, m_6 \ddot{X}_{cg6}, m_6 \ddot{Y}_{cg6} + m_6 g, \\ I_6 \ddot{\theta}_{cg6}, 0, 0, 0, m_{slider} a_{slider} + m_{slider} g - Q \end{bmatrix}^T$$

Results of the Kinetostatic Analysis  
of  
Hybrid Driven Mechanism

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*** System inputs ***

s      = 0.0024 (m)
sdot= 0 (m/s)
sddot= 0 (m/s^2)
theta5 = 4.86 (rad)
w5= 1.0472 (rad/s)
alpha5 =0 (rad/s^2)
Q      = 100000 (Newton) (Forming force of the slider)

***Results ***

Fax = -6240.21 (N)
Fay = 3308.17 (N)
Fa = 7062.88 (N)
Angle of Fa = 152.07 (degrees)

Fbx = -6107.53 (N)
Fby = 2972.84 (N)
Fb = 6792.62 (N)
Angle of Fb = 154.046 (degrees)
F3x = -4422.43 (N)
F3y = 5899.67 (N)
F3 = 7373.2 (N)
Angle of F3 = 126.855 (degrees)

F4x = 3566.28 (N)
F4y = -99645.3 (N)
F4 = 99709.1 (N)
Angle of F4 = -87.9503 (degrees)

Fcx = 1841.99 (N)
Fcy = -99803.8 (N)
Fc = 99820.8 (N)
Angle of Fc = -88.9427 (degrees)

Ffx = -676.921 (N)
Ffy = -105374 (N)
Ff = 105376 (N)
Angle of Ff = 269.632 (degrees)

Fex = 676.88 (N)
Fey = 105400 (N)
Fe = 105402 (N)
Angle of Fe = 89.6321 (degrees)

F6x = -856.154 (N)
F6y = -105545 (N)
F6 = 105548 (N)
Angle of F6 = 269.535 (degrees)

T2 = -988.72 (Nm)
T5 = -3216.83 (Nm)
N = 1841.99 (N)

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Figure 7 Results of analysis

$D$  is a known matrix. Some components of the  $D$  matrix are given in a symbolic form.  $E$  is an unknown matrix including the reaction forces in all joints and torques in the cranks of the mechanism.  $F$  is a known matrix formed by inertia torques, inertia forces and an external force. Equations in [1] are used to obtain  $F$  matrix.  $E$  matrix can easily be found by an inverse operation. It is illustrated in Eq. (51).

$$E = D^{-1} \times F \quad (51)$$

The elements given in  $D$  matrix are;

$$D_{3,1} = r_{Acg2} \sin \theta_2, D_{3,2} = -r_{Acg2} \cos \theta_2, D_{3,3} = r_{Bcg2} \sin \theta_2, D_{3,4} = -r_{Bcg2} \cos \theta_2 \quad (52)$$

$$D_{6,3} = r_{Bcg3} \sin \theta_3, D_{6,4} = -r_{Bcg3} \cos \theta_3, D_{6,5} = r_{Dcg3} \sin \theta_3, D_{6,6} = -r_{Dcg3} \cos \theta_3 \quad (53)$$

$$D_{9,7} = r_{Dcg4} \sin \theta_4, D_{9,8} = r_{Dcg4} \cos \theta_4, D_{9,9} = r_{Ccg4} \sin \theta_4, D_{9,10} = r_{Ccg4} \cos \theta_4 \quad (54)$$

$$D_{12,11} = r_{Fcg5} \sin \theta_5, D_{12,12} = -r_{Fcg5} \cos \theta_5, D_{12,13} = -r_{Ecgs} \sin \theta_5, D_{12,14} = r_{Ecgs} \cos \theta_5 \quad (55)$$

$$D_{15,13} = -r_{Ecgs} \sin \theta_6, D_{15,14} = r_{Ecgs} \cos \theta_6, D_{15,15} = -r_{Dcg6} \sin \theta_6, D_{15,16} = r_{Dcg6} \cos \theta_6 \quad (56)$$

## 6.1 An Example Analysis

The reaction forces in each joint and the cranks torques can be obtained as illustrated below by using a Matlab® code written for kinetostatic equations. The data including the slider displacement, velocity and acceleration and the angular displacement, velocity and acceleration of the crank driven by the constant velocity motor and forming force of the slider are given as input parameters. Fig. 7 shows the output of the code written in Matlab®.

## 7 CONCLUSION

A hybrid seven-bar press mechanism is explained with link optimization. Genetic algorithm is applied for finding optimum link lengths for the mechanism. The variations of the design variables are checked. They all operate in the operating limits of the press mechanism. One mechanism satisfying eight precision point is taken as an example and presented here. Future work will include more precision points for performing the required motion. An example for kinetostatic analysis showing all forces in each link and torques in the cranks of the hybrid system is presented. Studies on this subject are continuing in Mechatronics laboratory.

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